BUREAU VERITAS July 2018
Rules for the Classification of Steel Ships

PART C – Machinery, Electricity, Automation and Fire Protection

ALIGNMENT

PT C, Ch 1 Sec 7
2.4 Stern tube bearings
2.4.1 Oil lubricated aft bearings of antifriction metal
a) The length of bearings lined with white metal or other antifriction metal is to be not less than twice the rule diameter of the shaft in way of the bearing.
b) The length of the bearing may be less than that given in (a) above, provided the nominal bearing pressure is not more than 0,8 N/mm^2, as determined by static bearing reaction calculations taking into account shaft and propeller weight, as exerting solely on the aft bearing, divided by the projected area of the shaft.
However, the minimum bearing length is to be not less than 1,5 times its actual inner diameter.
2.4.2 Oil lubricated aft bearings of synthetic rubber, reinforced resin or plastics material
a) For bearings of synthetic rubber, reinforced resin or plastics material which are approved by the Society for use as oil lubricated sternbush bearings, the length of the bearing is to be not less than twice the rule diameter of the shaft in way of the bearing.
b) The length of the bearing may be less than that given in (a) above provided the nominal bearing pressure is not more than 0,6 N/mm^2, as determined according to [2.4.1] b).
However, the minimum length of the bearing is to be not less than 1,5 times its actual inner diameter.
Where the material has proven satisfactory testing and operating experience, consideration may be given to an increased bearing pressure.
2.4.3 Water lubricated aft bearings of lignum vitae or antifriction metal
Where the bearing comprises staves of wood (known as "lignum vitae") or is lined with antifriction metal, the length of the bearing is to be not less than 4 times the rule diameter of the shaft in way of the bearing.
2.4.4 Water lubricated aft bearings of synthetic materials
a) Where the bearing is constructed of synthetic materials which are approved by the Society for use as water lubricated sternbush bearings, such as rubber or plastics, the length of the bearing is to be not less than 4 times the rule diameter of the shaft in way of the bearing.
b) For a bearing design substantiated by experimental data to the satisfaction of the Society, consideration may be given to a bearing length less than 4 times, but in no case less than 2 times, the rule diameter of the shaft in way of the bearing.
2.4.5 Grease lubricated aft bearings
The length of grease lubricated bearings is generally to be not less than 4 times the rule diameter of the shaft in way of the bearing.

3.3 Shaft alignment for ships granted with a notation ESA
3.3.1 Application
Ships having the additional service feature or additional class notation **ESA**, as described respectively in Pt A, Ch 1, Sec 2, [4.1.5] and in Pt A, Ch 1, Sec 2, [6.14.31], are to comply with the requirements of Rule Note NR592 Elastic Shaft Alignment.
3.4 Shaft alignment for ships not granted with a notation ESA
3.4.1 General
For ships to which the notation **ESA** is not assigned, shaft alignment calculations and shaft alignment procedures are to be submitted for review in the following cases:
• propulsion plants with a shaft diameter of 300 mm or greater in way of the aftermost bearing, whether or not they comprise a gearbox
• geared propulsion plants with power take-in (PTI) or power take-off (PTO), where the shaft diameter in way of the aftermost bearing is 150 mm or greater.
3.4.2 Shaft alignment calculations

a) Scope of the calculations
The shaft alignment calculations are to be carried out in the following conditions:
1) alignment conditions during the shafting installation (ship in dry dock or afloat with propeller partly immersed)
2) cold, static, afloat conditions
3) hot, static, afloat conditions
4) hot, running conditions.
Note 1: Vertical and horizontal calculations are to carried out, as deemed relevant.

b) Information to be submitted
The shaft alignment calculation report should contain the following information:
1) Description of the shaftline model:
• length, diameters and density of material for each shaft

• definition of the reference line
• longitudinal, vertical and horizontal position of the bearing with respect to the reference line
• bearings characteristics: material, length, clearance.

2) Input parameters
• hydrodynamic propeller loads (horizontal and vertical forces and moments)
• weight and buoyancy effect of the propeller, depending on the propeller immersion corresponding to the different loading cases of the ship
• engine power and rotational speed of the propeller (for calculations in running conditions)
• machining data of aft bush slope boring
• for slow speed engines, equivalent model of the crankshaft, with indication of the input loads
• for geared installation, gear tooth forces and moments
• thermal expansion of the gearbox or of the main engine between cold and hot conditions
• jack-up location.

3) Limits
• limits specified by engine or gearbox manufacturer (such as allowable bearing loads, allowable moments and forces at the shaft couplings)
• allowable loads specified by bearing manufacturer.

4) Results
• bearings influence coefficients table
• expected bearing reactions, for the different calculation conditions
• expected shaft deflections, shear forces and bending moments alongside the shaftline, for the different calculation conditions
• gap and sag values (depending on the alignment method)
• jack-up correction factors.

c) Acceptability criteria for the calculations
The results of the shaft alignment calculations are to comply with the following acceptability criteria:
• Relative slope between propeller shaft and aftermost boring axis is not to exceed 0,3 mm/m
• all bearings are to remain loaded
• loads on intermediate shaft bearings are not to exceed 80% of the maximum permissible load specified by the manufacturer
• stern tube bearing loads are not to exceed the limits started in [2.4].

3.4.3 Shaft alignment procedure
The shaft alignment procedure is to be submitted for review and is to be consistent with the shaft alignment calculations.
The shaft alignment procedure should include at least the following:
• Ship conditions in which the alignment is to be carried out (drafts, propeller immersion, engine room temperature)
• Method used for establishing the reference line (using laser or optical instruments or piano wire)
• Description of the different steps of the shafting installation:
- bearing slope boring
- setting of the bearing offset and installation of the temporary shaft support (where relevant) in accordance with the results of the shaft alignment calculation
- flange coupling parameter setting (gap and sag)
- bearing load test (jack-up test).

3.4.4 Verification of the alignment procedure
The purpose of the verification procedure is to ensure that the alignment measurements comply with the calculated values. The shaft alignment verification procedure is described in Ch 1, Sec 15, [3.11.1].
The shaft alignment verification is to be carried out by the shipyard in presence of the Surveyor and submitted to the Society.
The criteria for acceptability of the alignment conditions include the following:
• the position of the two aftermost bearings should not differ from the specified offsets by more than ± 0,1 mm.
• Gap and sag values should not differ from the calculated values by more than ± 0,1 mm.
• Bearing loads should not differ from the calculated values by more than ± 20%.

TORSIONAL VIBRATION

PT C, Ch 1, Sec 9

1.1 Application

1.1.1 The requirements of this Section apply to the shafting of the following installations:
• propulsion systems with prime movers developing 220 kW or more
• other systems with internal combustion engines developing 110 kW or more and driving auxiliary machinery intended for essential services.

1.1.2 Exemptions

The requirements of this Section may be waived in cases where satisfactory service operation of similar installations is demonstrated.

2 Design of systems in respect of vibrations

2.1 Principle

2.1.1 General

a) Special consideration shall be given by Manufacturers *to the design, construction and installation of propulsion machinery systems so that any mode of their vibrations shall not cause undue stresses in these systems in the normal operating ranges.*

b) Calculations are to be carried out for the configurations of the system likely to have influence on the torsional vibrations.

c) Where deemed necessary by the Manufacturer, axial and/or bending vibrations are to be investigated.

2.1.2 Vibration levels

Systems are to have torsional, bending and axial vibrations both in continuous and in transient running acceptable to the Manufacturers, and in accordance with the requirements of this section.

Where vibrations are found to exceed the limits stated in this Section, the builder of the plant is to propose corrective actions, such as:

• operating restrictions, provided that the owner is informed, or

• modification of the plant.

2.1.3 Condition of components

Systems are to be designed considering the following conditions, as deemed necessary by the Manufacturer:

• engine: cylinder malfunction

• flexible coupling: possible variation of the stiffness or damping characteristics due to heating or ageing

• vibration damper: possible variation of the damping coefficient.

2.2 Modifications of existing plants

2.2.1 Where substantial modifications of existing plants,such as:

• change of the running speed or power of the engine

• replacement of an important component of the system (propeller, flexible coupling, damper) by one of different characteristics, or

• connection of a new component are carried out, new vibration analysis is to be submitted for approval.

3 Torsional vibrations

3.1 Documentation to be submitted

3.1.1 Calculations

Torsional vibration calculations are to be submitted for the various configurations of the plants, showing:

• the equivalent dynamic system used for the modeling of the plant, with indication of:

- inertia and stiffness values for all the components of the system

- outer and inner diameters and material properties of the shafts

• the natural frequencies

• the values of the vibratory torques or stresses in the components of the system for the most significant critical speeds and their analysis in respect of the Rules and other acceptance criteria

• the possible restrictions of operation of the plant.

3.1.2 Particulars to be submitted

The following particulars are to be submitted with the torsional vibration calculations:

a) for turbines, multi-engine installations or installations with power take-off systems:

• description of the operating configurations

• load sharing law between the various components for each configuration

b) for installations with controllable pitch propellers, the power/rotational speed values resulting from the combinatory operation

c) for prime movers, the service speed range and the minimum speed at no load

d) for internal combustion engines:

• manufacturer and type

• nominal output and rotational speed

• mean indicated pressure

• number of cylinders

• “V” angle

• firing angles

• bore and stroke

• excitation data, such as the polynomial law of harmonic
components of excitations

• nominal alternating torsional stress considered for crankpin and journal

Note 1: The nominal alternating torsional stress is part of the basic data to be considered for the assessment of the crankshaft. It is defined in Ch 1, App 1.

e) for turbines:

• nominal output and rotational speed

• power/speed curve and range of operation

• number of stages, and load sharing between the stages

• main excitation orders for each rotating disc

• structural damping of shafts

• external damping on discs (due to the fluid)

f) for reduction or step-up gears, the speed ratio for each step

g) for flexible couplings:
• the maximum torque
• the nominal torque
• the permissible vibratory torque
• the permissible heat dissipation
• the relative damping
• the torsional dynamic stiffness / transmitted torque relation where relevant
h) for torsional vibration dampers:
• the manufacturer and type
• the permissible heat dissipation
• the damping coefficient
• the inertial and stiffness properties, as applicable
i) for propellers:
• the type of propeller: ducted or not ducted
• the number of propellers of the ship
• the number of blades
• the excitation and damping data, if available
j) for electric motors, generators and pumps, the drawing of the rotating parts, with their mass moment of inertia and main dimensions.

3.2 Definitions, symbols and units

3.2.1 Definitions

a) Torsional vibration stresses referred to in this Article are the stresses resulting from the alternating torque corresponding to the synthesis of the harmonic orders concerned.
b) The misfiring condition of an engine is the malfunction of one cylinder due to the absence of fuel injection (which results in a pure compression or expansion in the cylinder).

3.2.2 Symbols, units

The main symbols used in this Article are defined as follows:

τ : Torsional vibration stress, as defined in [3.2.1], in N/mm^2

τ_1 : Permissible stress due to torsional vibrations for continuous operation, in N/mm^2

τ_2 : Permissible stress due to torsional vibrations for transient running, in N/mm^2

R_m : Tensile strength of the shaft material, in N/mm^2

C_R : Material factor, equal to:

$$\frac{R + 160}{18}$$

d : Minimum diameter of the shaft, in mm

C_D : Size factor of the shaft, equal to:

$$0,35 + 0,93\, d^{-0,2}$$

N : Speed of the shaft for which the check is carried out, in rev/min

N_n : Nominal speed of the shaft, in rev/min

N_c : Critical speed, in rev/min

λ : Speed ratio, equal to N/N_n

C_λ : Speed ratio factor, equal to:

- $3 - 2\lambda^2$ for $\lambda < 0,9$
- $1,38$ for $0,9 \le \lambda < 1,05$

C_k : Factor depending on the stress concentration factor of the shaft design features given in Tab 1.

Table 1 : Values of C_k factors

Intermediate shafts with							Thrust shafts external to engines		Propeller shafts		
straight sections and integral coupling flanges	shrink-fit couplings (1)	keyways, tapered connection (2)	keyways, cylindrical connection (2)	radial hole	longitudinal slot (3)	splined shafts	on both sides of thrust collar	in way of axial bearing where a roller bearing is used as a thrust bearing	flange mounted or keyless fitted propellers (4)	key fitted propellers (4)	between forward end of aft most bearing and forward stern tube seal
1,00	1,00	0,60	0.45	0.50	0.30	0,80	0,85	0,85	0,55	0,55	0.80

(1) C_k values refer to the plain shaft section only. Where shafts may experience vibratory stresses close to the permissible stresses for continuous operation, an increase in diameter to the shrink fit diameter is to be provided, e.g. a diameter increase of 1 to 2 % and a blending radius as described in Ch 1, Sec 7, [2.5.1]

(2) Keyways are to be in accordance with the provisions of Ch 1, Sec 7, [2.5.1].

(3) Subject to limitations as $\ell/d_o < 0,8$ and $d_i/d_o < 0,7$ and $e/d_o > 0,15$, where:

ℓ : Slot length in mm

e : Slot width in mm

d_i, d_o : As per Sec 7, [2.2.3]

The C_k value is valid for 1, 2 and 3 slots, i.e. with slots at 360 respectively 180 and respectively 120 degrees apart.

(4) Applicable to the portion of the propeller shaft between the forward edge of the aftermost shaft bearing and the forward face of the propeller hub (or shaft flange), but not less than 2.5 times the required diameter.

Note 1: Higher values of C_k factors based on direct calculations may also be considered.

Note 2: The determination of C_k factors for shafts other than those given in this table will be given special consideration by the Society.

3.3 Calculation principles

3.3.1 Method

a) Torsional vibration calculations are to be carried out using a recognised method.

b) Where the calculation method does not include harmonic synthesis, attention is to be paid to the possible superimposition of two or more harmonic orders of different vibration modes which may be present in some restricted ranges.

3.3.2 Scope of the calculations

a) Torsional vibration calculations are to be carried out considering:

- normal firing of all cylinders, and
- misfiring of one cylinder.

b) Where the torsional dynamic stiffness of the coupling depends on the transmitted torque, two calculations are to be carried out:

- one at full load
- one at the minimum load expected in service.

c) For installations with controllable pitch propellers, two calculations are to be carried out:

- one for full pitch condition
- one for zero pitch condition.

d) The calculations are to take into account other possible sources of excitation, as deemed necessary by the Manufacturer.

Electrical sources of excitations, such as static frequency converters, are to be detailed. The same applies to transient conditions such as engine start up, reversing, clutching in, as necessary.

e) The natural frequencies are to be considered up to a value corresponding to 15 times the maximum service speed. Therefore, the excitations are to include harmonic orders up to the fifteenth.

3.3.3 Criteria for acceptance of the torsional vibration loads under normal firing conditions

a) Torsional vibration stresses in the various shafts are not to exceed the limits defined in [3.4]. Higher limits calculated by an alternative method may be considered, subject to special examination by the Society.

The limit for continuous running τ_1 may be exceeded only in the case of transient running in restricted speed ranges, which are defined in [3.4.5]. In no case are the torsional vibration stresses to exceed the limit for transient running τ_2.

Propulsion systems are to be capable of running continuously without restrictions at least within the speed range between 0,8 N_n and 1,05 N_n. Transient running may be considered only in restricted speed ranges for speed ratios $\lambda \leq 0,8$.

Auxiliary machinery is to be capable of running continuously without restrictions at least within the range between 0,95 N_n and 1,1 N_n. Transient running may be considered only in restricted speed ranges for speed ratios $\lambda \leq 0,95$.

b) Torsional vibration levels in other components are to comply with the provisions of [3.5].

3.3.4 Criteria for acceptance of torsional vibration loads under misfiring conditions

a) The provisions of [3.3.3] related to normal firing conditions also apply to misfiring conditions.

Note 1: For propulsion systems operated at constant speed, restricted speed ranges related to misfiring conditions may be accepted for speed ratios $\lambda > 0,8$.

b) Where calculations show that the limits imposed for certain components may be exceeded under misfiring conditions, a suitable device is to be fitted to indicate the occurrence of such conditions.

3.4 Permissible limits for torsional vibration stresses in crankshaft, propulsion shafting and other transmission shafting

3.4.1 General

a) The limits provided below apply to steel shafts. For shafts made of other material, the permissible limits for torsional vibration stresses will be determined by the Society after examination of the results of fatigue tests carried out on the material concerned.

b) These limits apply to the torsional vibration stresses as defined in [3.2.1]. They relate to the shaft minimum section, without taking account of the possible stress concentrations.

3.4.2 Crankshaft

a) Where the crankshaft has been designed in accordance with Ch 1, App 1, the torsional vibration stresses in any point of the crankshaft are not to exceed the following limits:

- $\tau_1 = \tau_N$ for continuous running
- $\tau_2 = 1{,}7\ \tau_N$ for transient running,

where τ_N is the nominal alternating torsional stress on which the crankshaft scantling is based (see Note 1 in [3.1.2]).

b) Where the crankshaft has not been designed in accordance with Ch 1, App 1, the torsional vibration stresses in any point of the crankshaft are not to exceed the following limits:

- $\tau_1 = 0{,}55 \cdot C_R \cdot C_D \cdot C_\lambda$ for continuous running
- $\tau_2 = 2{,}3\ \tau_1$ for transient running.

3.4.3 Intermediate shafts, thrust shafts and propeller shafts

The torsional vibration stresses in any intermediate, thrust and propeller shafts are not to exceed the following limits:

- $\tau_1 = C_R \cdot C_k \cdot C_D \cdot C_\lambda$ for continuous running
- $\tau_2 = 1{,}7\ \tau_1 \cdot C_k^{-0{,}5}$ for transient running.

3.4.4 Transmission shafting for generating sets and other auxiliary machinery

The torsional vibration stresses in the transmission shafting for generating sets and other auxiliary machinery, such as pumps or compressors, are not to exceed the following limits:

- $\tau_1 = 0{,}90 \cdot C_R \cdot C_D$ for continuous running
- $\tau_2 = 5{,}4\ \tau_1$ for transient running.

3.4.5 Restricted speed ranges

a) Where the torsional vibration stresses exceed the limit τ_1 for continuous running, restricted speed ranges are to be imposed which are to be passed through rapidly.

b) The limits of the restricted speed range related to a critical speed N_c are to be calculated in accordance with the following formula:

$$\frac{16 \cdot N_c}{18 - \lambda} \le N \le \frac{(18 - \lambda) \cdot N_c}{16}$$

c) Where the resonance curve of a critical speed is obtained from torsional vibration measurements, the restricted speed range may be established considering the speeds for which the stress limit for continuous running τ_1 is exceeded.

d) Where restricted speed ranges are imposed, they are to be crossed out on the tachometers and an instruction plate is to be fitted at the control stations indicating that:

- the continuous operation of the engine within the considered speed range is not permitted
- this speed range is to be passed through rapidly.

e) When restricted speed ranges are imposed, the accuracy of the tachometers is to be checked in such ranges as well as in their vicinity.

f) Restricted speed ranges in one-cylinder misfiring conditions of single propulsion engine ships are to enable safe navigation.

3.5 Permissible vibration levels in components other than shafts

3.5.1 Gears

a) The torsional vibration torque in any gear step is not to exceed 30% of the torque corresponding to the approved rating throughout the service speed range.

Where the torque transmitted at nominal speed is less than that corresponding to the approved rating, higher torsional vibration torques may be accepted, subject to special consideration by the Society.

b) Gear hammering induced by torsional vibration torque reversal is not permitted throughout the service speed range, except during transient running at speed ratios $\lambda \le 0{,}3$. Where calculations show the existence of torsional vibration torque reversals for speed ratios $\lambda > 0{,}3$, the corresponding speed ranges are to be identified by appropriate investigations during sea trials and considered as restricted speed ranges in accordance with [3.4.5].

3.5.2 Generators

a) In the case of alternating current generators, the torsional vibration amplitude at the rotor is not to exceed $\pm 2{,}5$ electrical degrees at service rotational speed under full load working conditions.

b) Vibratory inertia torques due to torsional vibrations and imposed on the rotating parts of the generator are not to exceed the values M_A, in Nm, calculated by the following formulae, as appropriate:

- for $0{,}95 \le \lambda \le 1{,}1$: $M_A = \pm 2{,}5\ M_T$
- for $\lambda \le 0{,}95$: $M_A = \pm 6\ M_T$

where:

M_T : Mean torque transmitted by the engine under full load running conditions, in Nm

Note 1: In the case of two or more generators driven by the same engine, the portion of M_T transmitted to each generator is to be considered.

λ : Speed ratio defined in [3.2.2].

3.5.3 Flexible couplings

a) Flexible couplings are to be capable of withstanding the mean transmitted torque and the torsional vibration torque throughout the service speed range, without exceeding the limits for continuous operation imposed by the manufacturer (permissible vibratory torque and power loss).

Where such limits are exceeded under misfiring conditions, appropriate restrictions of power or speed are to be established.

b) Flexible couplings fitted in generating sets are also to be capable of withstanding the torques and twist angles arising from transient criticals and short-circuit currents. Start up conditions are also to be checked.

3.5.4 Dampers

a) Torsional vibration dampers are to be such that the permissible power loss recommended by the manufacturer is not exceeded throughout the service speed range.

b) Dampers for which a failure may lead to a significant vibration overload of the installation will be the subject of special consideration.

3.6 Torsional vibration measurements

3.6.1 General

a) The Society may require torsional vibration measurements to be carried out under its attendance in the following cases:

- where the calculations indicate the possibility of dangerous critical speeds in the operating speed range
- where doubts arise as to the actual stress amplitudes or critical speed location, or
- where restricted speed ranges need to be verified.

b) Where measurements are required, a comprehensive report including the analysis of the results is to be submitted to the Society.

3.6.2 Method of measurement

When measurements are required, the method of measurement is to be submitted to the Society for approval. The type of measuring equipment and the location of the measurement points are to be specified.

BENDING AND AXIAL VIBRATION

See [2.1.1 c)]